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FRICTION FACTORS FOR HELICAL CORRUGATED PIPE

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FRICTION FACTORS FOR HELICAL CORRUGATED PIPE[1]

3a [by

Charles E. Rice[2]

INTRODUCTION

Accurate values of the Darcy-Weisbach friction factor for 8-inch and 12-inch-diameter helical corrugated pipe were needed in order to compute true values of the entrance losses for hooded entrance (2)[3] pipe spillways being tested at the Stillwater Outdoor Hydraulic Laboratory of the Agricultural Research Service. Available information on friction factors for helical corrugated pipe included values given in a manufacturer's handbook (1) and the data of Chamberlain who tested a 12-inch-diameter pipe (4). These data are useful for design purposes, but the need for accurate values of the friction factor for the particular pipes being investigated required the experiments described in this report.

NOTATION

The following symbols have been adopted for use in this publication:

A = area of pipe = $\pi D^2/4$
D = nominal diameter of pipe
f = Darcy-Weisbach friction factor
g = acceleration due to gravity
h_f = head loss
h_v = velocity head in pipe = $V^2/2g$
ℓ = conduit length
n = Manning roughness coefficient
Q = discharge
$\mathbb{R}$ = Reynolds number
r = corrugation depth, outside
S = slope of hydraulic grade line = h_f/ℓ
t = pipe wall thickness
V = velocity in pipe = Q/A
x = corrugation pitch
y = elevation above the outlet invert where hydraulic grade line intersects the plane of the conduit exit
z = corrugation depth, inside
β = proportion of the pipe diameter above the outlet invert at which the hydraulic grade line pierces the plane of the conduit exit = y/D
ϕ = helix angle of corrugation
ν = kinematic viscosity.

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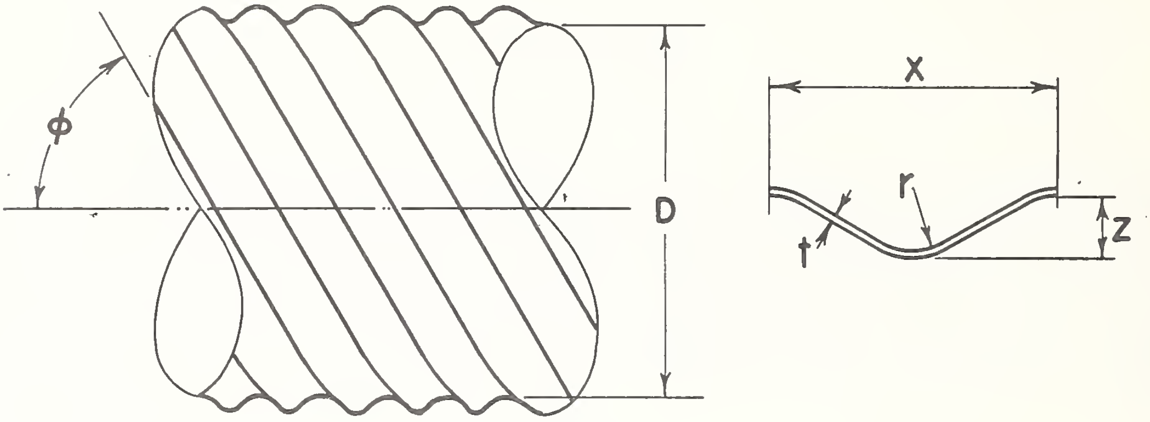
[3] Underscored numbers in parentheses refer to Literature at end of report.

THE EXPERIMENT

Helical pipe has corrugations and a continuous lock seam that run helically around the pipe. Boundary dimensions for the pipes tested are presented in figure 1. These dimensions are the average of 10 individual measurements made at randomly selected points on each test pipe.



Nominal Pipe Dia. in.	Actual Pipe Dia. in.	ϕ deg.	r in.	t in.	x 1/ in.	z in.	D/z
8	8.12	62	0.24	0.07	1.50	0.24	33
12	12.05	66	0.43	0.07	2.00	0.43	28

1/ Measured at right angles to corrugation.

Figure 1.--Pipe boundary dimensions.

The helical pipes tested were of standard manufacture and were obtained new from commercial sources. Four 20-foot pipes with flanged ends were bolted together to make an 80-foot test length. This test pipe was supported in a horizontal position, aligned with a transit, and tied to the supports to prevent movement. A sketch and general view of the test setup are presented in figures 2 and 3, respectively.

Eight static tubes, centered in the pipe cross section and located along the pipe at approximately 10-foot intervals, were used to measure the pressure gradient at the center of the pipe cross section in the test pipes. These tubes were equivalent to the static leg of the standard Prandtl pitot tube (6). A schematic diagram of the static tube is shown in figure 2, detail B. The tubes were constructed of 3/8-inch-outside-diameter, hard drawn brass tubing. The upstream tip of the tube was a hemisphere. Four piezometer holes were equally spaced around the circumference of the tube with the hole centerlines three tube diameters downstream from the extreme upstream point of the tip. The ratio of the piezometer hole diameter to the tube wall thickness was 0.8.

For the first few tests the static tubes were connected to 5/16-inch-inside-diameter glass tubes mounted on individual manometer boards at each static tube location as shown in figure 3.

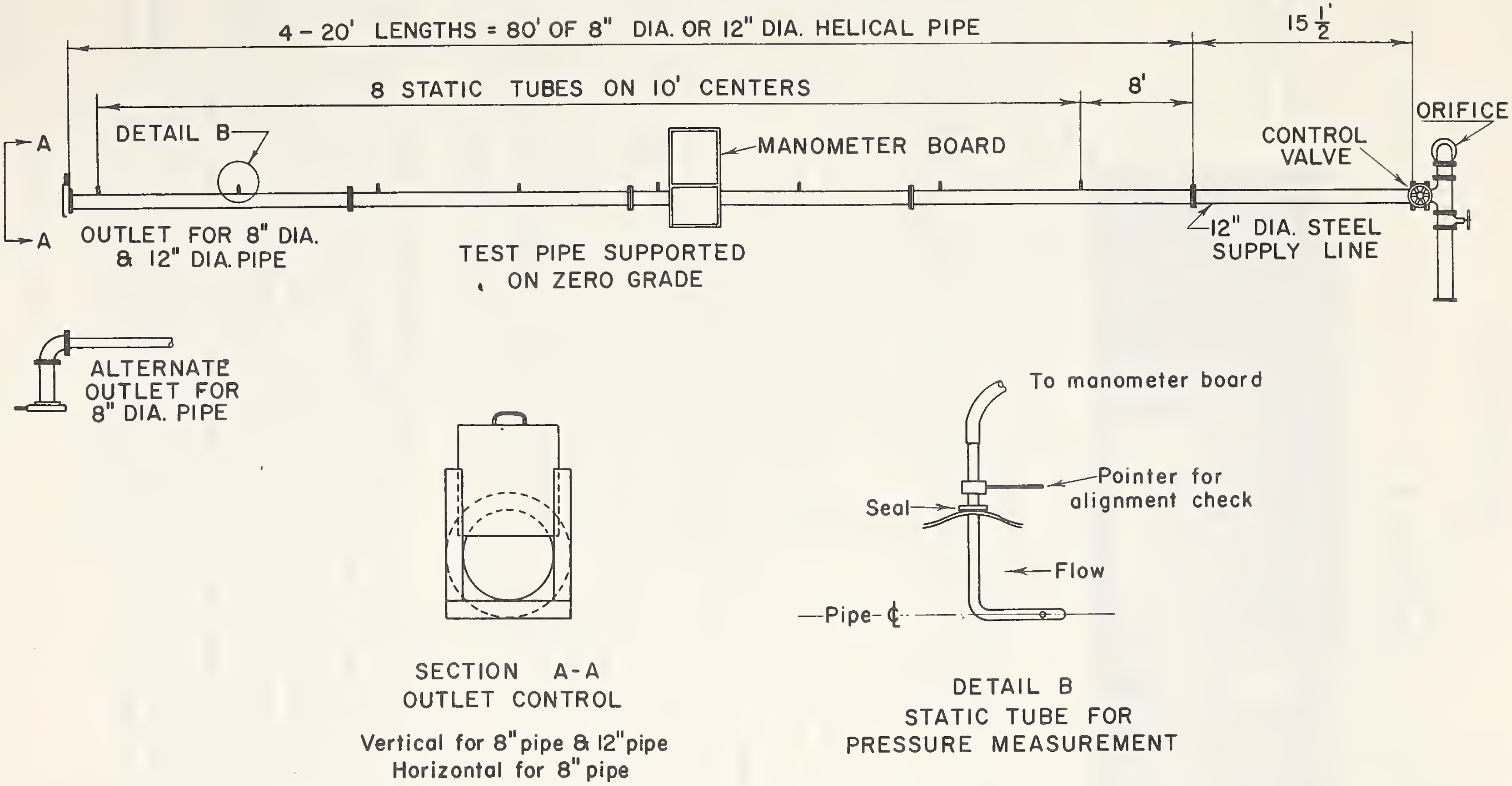

4 - 20' LENGTHS = 80' OF 8" DIA. OR 12" DIA. HELICAL PIPE
15 1/2'
8 STATIC TUBES ON 10' CENTERS
8'
ORIFICE
CONTROL VALVE
DETAIL B
MANOMETER BOARD
A
A
OUTLET FOR 8" DIA. & 12" DIA. PIPE
TEST PIPE SUPPORTED ON ZERO GRADE
12" DIA. STEEL SUPPLY LINE
ALTERNATE OUTLET FOR 8" DIA. PIPE
To manometer board
Pointer for alignment check
Seal
Flow
Pipe-℄
SECTION A-A
OUTLET CONTROL
Vertical for 8" pipe & 12" pipe
Horizontal for 8" pipe
DETAIL B
STATIC TUBE FOR
PRESSURE MEASUREMENT

Figure 2.--Arrangement for friction factor determination for 8-inch and 12-inch helical corrugated pipe.

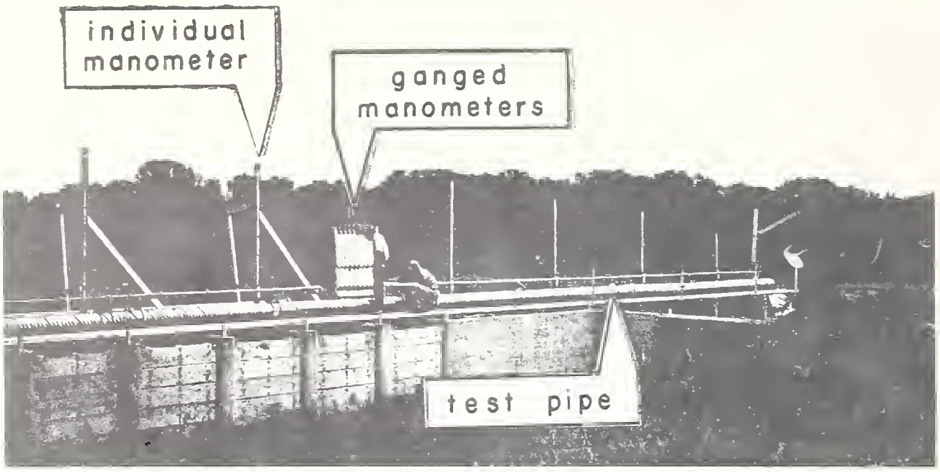


Figure 3.--General view of test setup.

A scale, graduated in hundredths of a foot, was placed behind the glass tubes. The water level was read by eye to the nearest hundredth of a foot.

To improve the precision of the manometer measurements the simple manometers were replaced with 1-1/2-inch-diameter glass tubes and point gages which could be read to the nearest 0.001 foot. These tubes were grouped on one frame, as shown in figure 3, so they could be coupled to a common header pipe for the simultaneous zeroing of the eight point gages with a common water level. This was a significant improvement over the zeroing of the individual manometers which were zeroed by using an engineer's level. Although the grouping of the manometers on one frame greatly reduced zeroing errors, it did require long hose lines from the static tubes to the manometers with attendant air bubble trouble. The use of clear plastic tubing for these lines reduced the bubble problem in that the lines could be inspected frequently. If bubbles were present, they could be easily bled out of the lines.

For a second series of tests made on the 8-inch pipe in 1964, about 13 months after the first series of tests, piezometer openings were made in the wall of the test pipe to obtain pressures at the periphery of the pipe for comparison with the pressures measured at the center of the pipe cross section. These piezometers were located at the same stations along the pipe as the piezometers in the static tubes. The piezometer openings were drilled in the pipe wall from the outside and were not accessible for inspection. However, an attempt was made to remove any burrs that might have been present by inserting in the hole a small stiff wire with a 90-degree bend, and rotating it to smooth the surface. The piezometer openings were placed in the trough of a corrugation relative to the inside of the pipe. The ratio of the piezometer hole diameter to pipe wall thickness was about 1.6.

All tests were run with full pipe flow. Since the test pipes were horizontal, a restriction was required to force the pipes to flow full at the lower discharges. For the 12-inch pipe this restriction was a vertical sliding gate attached to the outlet end flange as shown in section A-A, figure 2. For the first tests on the 8-inch pipe the restriction was a horizontal sliding gate at the bottom of a vertical downspout attached to the test pipe with a 12-inch, 90-degree elbow. This is shown in figure 2 labeled "Alternate outlet for 8-inch-diameter pipe." For the 1964 tests on the 8-inch pipe the restriction was a vertical sliding gate identical to that used with the 12-inch pipe tests.

The discharge for the tests was measured with sharp-edged orifices in the supply line. The orifices had been calibrated in place. An air-water differential manometer was used to measure the head drop across the orifice. Discharges were determined from orifice calibration tables using the arithmetic average of 20 individual readings of the head drop across the orifice.

TEST PROCEDURE

The flow through the test pipe was adjusted to the desired rate by manipulating the upstream control valve and the adjustable gate at the outlet end until the correct head drop across the orifice was obtained. After the desired flow rate had been obtained the outlet gate was adjusted to place the hydraulic grade line within the range of the manometers. This fine adjustment of the outlet gate did not significantly affect the pre-set discharge. After the flow had steadied, a series of 10 instantaneous readings was recorded for each leg of the orifice manometer. Next, the hydraulic grade line manometer readings were obtained by reading manometers one through eight consecutively, at approximately 6-second intervals, until a total of 10 readings had been recorded for each. The orifice manometer readings were then repeated and the water temperature recorded. If the averages of the two sets of orifice manometer readings differed from each other by more than 1 percent, these test data were discarded and the test rerun. Otherwise, a new flow rate was set, time allowed for the flow to stabilize, and the test procedure repeated. After the maximum flow with the restricted outfall condition had been run, the restriction at the outlet was removed and tests were run with free outfall.

In the 1964 tests with the 8-inch pipe, one additional step was required. This was to obtain the pressure readings for the periphery piezometers. The grouped manometers and point gages were used to read these pressures by disconnecting the static tubes after the center pressures had been recorded and connecting the periphery piezometers in their place.

ANALYSIS

The friction factor was computed from the Darcy-Weisbach formula

$$h_f = f \frac{\ell V^2}{D2g} \qquad (1)$$

where h_f = head loss in feet

f = Darcy-Weisbach friction factor

ℓ = pipe length in feet

D = nominal pipe diameter in feet

V = average velocity in feet per second

g = acceleration of gravity in feet per second per second

The equation was reduced to the form

$$f = \frac{SD}{h_v} \qquad (2)$$

where $S = h_f/\ell$ = slope of the hydraulic grade line

$h_v = V^2/2g$ = velocity head in feet

The average velocity was obtained by dividing the discharge by the area corresponding to the nominal diameter of the pipe. The slope of the hydraulic grade line was calculated by the method of least squares using only those points in the zone of established flow. This zone was assumed to be in the portion of the pipe line having a uniform hydraulic grade line as indicated by a preliminary plotting.

The Reynolds number was computed by using the formula

$$\mathbb{R} = \frac{VD}{\nu} \tag{3}$$

$\mathbb{R}$ = Reynolds number

ν = Kinematic viscosity in square feet per second

The Manning coefficient n was computed from the f values by the relationship

$$n = \frac{f^{1/2} D^{1/6}}{13.6} \tag{4}$$

The elevation of the hydraulic grade line at the outlet end of the pipe was determined by extending the straight-line portion of the hydraulic grade line until it pierced the plane of the outlet end of the pipe. The vertical distance, y, from the piercing point to the pipe outlet invert was determined and the ratio

$$\beta = y/D \tag{5}$$

was calculated.

RESULTS

Summaries of the test results are presented in tables 1, 2, and 3.

The friction factor as a function of the Reynolds number is presented in figure 4 for the 12-inch and 8-inch pipes. For the 12-inch-diameter pipe the friction factor has no systematic variation with the Reynolds number in the flow range studied, but for the 8-inch-diameter pipe the friction factor decreases slightly with increasing Reynolds number.

The values of Manning's n listed in tables 1, 2, and 3 are nearly constant for each pipe. The average values of Manning's n for the 12-inch and 8-inch pipes are 0.015 and 0.013, respectively.

Friction factor values for the 8-inch-diameter pipe tests in 1964 computed by using the center pressures and the periphery pressures are in good agreement. These values are presented in table 3.

Figure 5 shows that β decreased rapidly as the discharge increased.

TABLE 1.--Summary of test results (12-inch helical corrugated pipe).

V	Q	S	Water temp.	IR	f	n	$Q/D^{5/2}$	β
F.p.s.	C.f.s.	Percent	Deg. F	10^{-4}			Ft.$^{1/2}$/sec.	
0.481	0.378	0.0163	68.0	4.43	0.0454	0.0157		
.626	.492	.0244	68.0	5.77	.0400	.0147		
.807	.634	.0388	68.0	7.44	.0383	.0144		
.933	.733	.0539	68.0	8.60	.0398	.0147		
1.065	.837	.0709	68.0	9.82	.0402	.0147		
1.154	.907	.0868	68.0	10.64	.0420	.0151		
1.114	.875	.0806	69.0	10.41	.0418	.0151		
1.261	.990	.1070	69.0	11.79	.0433	.0153		
1.428	1.122	.1340	72.0	13.89	.0423	.0151		
1.600	1.257	.1690	72.0	15.57	.0425	.0152		
1.834	1.440	.2210	72.0	17.84	.0423	.0151		
2.170	1.704	.3130	72.0	21.11	.0428	.0152		
2.556	2.007	.4150	72.0	24.86	.0409	.0149		
2.778	2.182	.4870	71.5	26.84	.0406	.0148		
3.212	2.523	.6590	71.5	31.04	.0411	.0149		
3.742	2.939	.8990	71.5	36.16	.0413	.0149		
4.324	3.396	1.2220	71.5	41.78	.0421	.0151		
4.970	3.904	1.6480	72.0	48.35	.0430	.0152		
4.164	3.270	1.0880	71.5	40.23	.0404	.0148		
5.821	4.572	2.2790	71.5	56.24	.0433	.0153		
6.371	5.004	2.7880	71.5	61.56	.0442	.0155		
6.967	5.472	3.4360	71.5	67.32	.0456	.0157		
7.192	5.649	3.6220	71.5	69.49	.0451	.0156		
7.434	5.839	3.6430	71.5	71.83	.0424	.0151	5.839	[1]0.592
7.190	5.647	3.3780	71.5	69.47	.0421	.0151	5.647	[1].617
6.663	5.233	2.8860	71.5	64.38	.0419	.0150	5.233	[1].717
5.969	4.688	2.2920	71.5	57.67	.0414	.0150	4.688	[1].767
5.039	3.958	1.6180	71.5	48.69	.0410	.0149	3.958	[1].854

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[1] Free outfall tests--remainder of tests with restricted outfall.

TABLE 2.--Summary of test results (8-inch helical corrugated pipe).

V	Q	S	Water temp.	$\mathbb{R}$	f	n	$Q/D^{5/2}$	β
F.p.s.	C.f.s.	Percent	Deg. F	10^{-4}			Ft.$^{1/2}$/sec.	
8.300	2.897	5.6590	75.5	56.23	0.0353	0.0129		
7.213	2.518	4.2980	75.5	48.87	.0355	.0129		
6.287	2.195	3.2940	76.0	42.86	.0358	.0130		
5.496	1.918	2.5120	76.0	37.47	.0357	.0130		
3.161	1.103	0.8330	75.0	21.28	.0358	.0130		
3.697	1.291	1.1430	75.0	24.90	.0359	.0130		
4.214	1.471	1.4800	75.0	28.38	.0358	.0130		
4.760	1.662	1.8920	75.0	32.05	.0359	.0130		
1.029	0.359	0.0988	75.5	6.97	.0400	.0137		
1.404	.490	.1840	76.0	9.57	.0400	.0137		
1.691	.590	.2610	76.0	11.53	.0391	.0136		
1.894	.661	.3280	75.5	12.83	.0393	.0136		
2.113	.738	.3800	75.0	14.23	.0366	.0131		
2.522	.880	.5360	75.0	16.98	.0361	.0131		
1.990	.695	.3490	73.5	13.16	.0378	.0134		
0.739	.258	.0540	73.5	4.89	.0425	.0142		
.900	.314	.0776	73.5	5.95	.0411	.0139		
10.021	3.498	8.1720	72.5	65.37	.0349	.0128		
11.223	3.918	10.1800	72.5	73.21	.0347	.0128		
11.346	3.960	10.3650	73.0	74.52	.0346	.0128	10.916	[1]0.055
10.652	3.718	9.2220	73.0	69.96	.0349	.0128	10.248	[1].078
9.656	3.371	7.6240	73.0	63.42	.0351	.0129	9.291	[1].198
8.827	3.081	6.3830	73.0	57.98	.0352	.0129	8.493	[1].294
7.378	2.575	4.4990	73.0	48.46	.0355	.0129	7.098	[1].426
5.466	1.908	2.5080	73.0	35.90	.0360	.0130	5.259	[1].693
0.452	0.158	0.0189	73.5	2.99	.0398	.0137		
.603	.210	.0361	73.0	3.96	.0426	.0142		

[1] Free outfall tests--remainder of tests with restricted outfall.

TABLE 3.--Summary of test results (1964, 8-inch helical corrugated pipe tests).

V	Q	S	Water temp.	$\mathbb{R}$	f	n	fp[1]	$Q/D^{5/2}$	β
F.p.s.	C.f.s.	Percent	Deg. F.	10^{-4}		Ft.$^{1/6}$		Ft.$^{1/2}$/sec.	
0.517	0.180	0.025	80	3.70	0.0402	0.0138	0.0415		
.653	.228	.038	82	4.79	.0383	.0134	.0425		
.765	.267	.052	82	5.61	.0381	.0134	.0416		
.855	.298	.071	81	6.20	.0418	.0140	.0394		
.979	.342	.092	77	6.75	.0410	.0139	.0409		
.951	.332	.081	81	6.84	.0383	.0134	.0401		
1.425	.497	.189	82	10.45	.0400	.0137	.0402		
1.810	.632	.304	82	13.27	.0399	.0137	.0403		
2.085	.728	.381	78	14.57	.0377	.0133	.0367		
2.387	.833	.485	79	16.90	.0365	.0131	.0367		
2.620	.915	.588	79	18.54	.0368	.0132	.0368		
2.382	.832	.490	81	17.26	.0371	.0132	.0370		
3.414	1.192	1.005	80	24.45	.0370	.0132	.0370		
4.360	1.522	1.629	80	31.22	.0368	.0132	.0371		
5.044	1.761	2.179	80	36.12	.0368	.0132	.0374		
5.649	1.972	2.817	82	41.43	.0379	.0134	.0378		
4.416	1.542	1.663	80	31.62	.0366	.0132	.0370		
6.128	2.139	3.211	80	43.88	.0367	.0132	.0368		
4.561	1.592	1.812	82	33.45	.0374	.0133	.0376		
6.496	2.268	3.658	82	47.64	.0372	.0132	.0377		
8.471	2.957	6.248	81	61.38	.0374	.0133	.0376		
9.751	3.404	8.129	81	70.66	.0367	.0132	.0375		
10.250	3.578	8.860	84	77.03	.0362	.0131	.0361		
9.513	3.320	7.596	84	71.49	.0360	.0130	.0373		
8.903	3.108	6.793	84	66.91	.0368	.0132	.0372		
8.141	2.842	5.595	82	59.70	.0363	.0131	.0367		
6.408	2.237	3.459	82	47.00	.0362	.0131	.0351	6.164	[2] 0.615
6.701	2.339	3.755	82	49.14	.0359	.0130	.0361	6.445	[2] .490
7.418	2.589	4.609	82	54.40	.0360	.0130	.0365	7.135	[2] .418
8.253	2.881	5.688	83	61.27	.0359	.0130	.0363	7.938	[2] .365
8.855	3.091	6.466	83	65.74	.0354	.0129	.0360	8.517	[2] .326
9.779	3.413	7.871	83	72.60	.0353	.0129	.0357	9.406	[2] .214
10.359	3.616	8.876	83	76.90	.0355	.0129	.0360	9.964	[2] .134

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[1] Friction factor calculated from periphery pressure measurements.
[2] Free outfall tests--remainder of tests with restricted outfall.

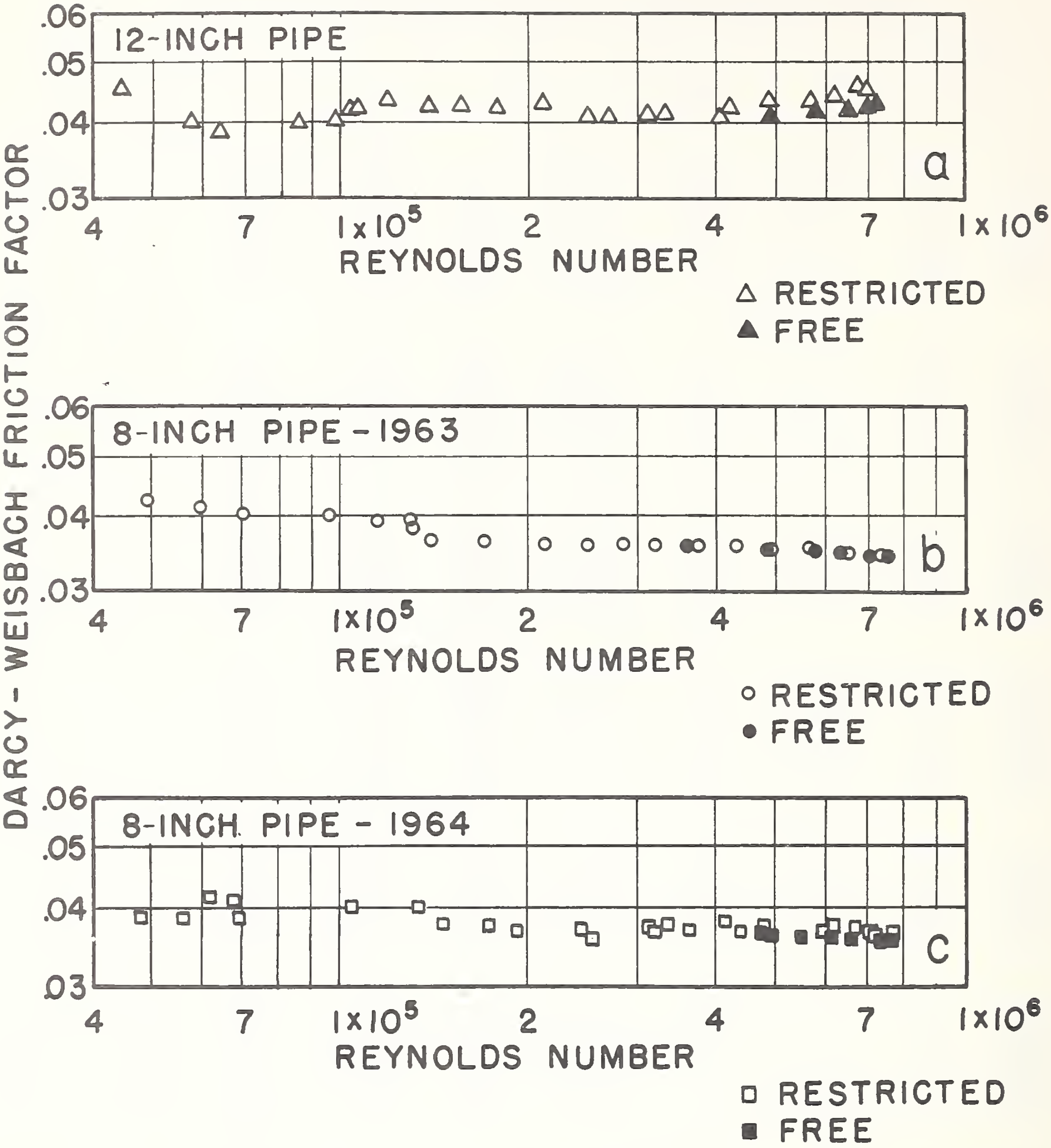

Figure 4.--Darcy-Weisbach friction factor versus Reynolds number for 8-inch and 12-inch helical corrugated metal pipe.

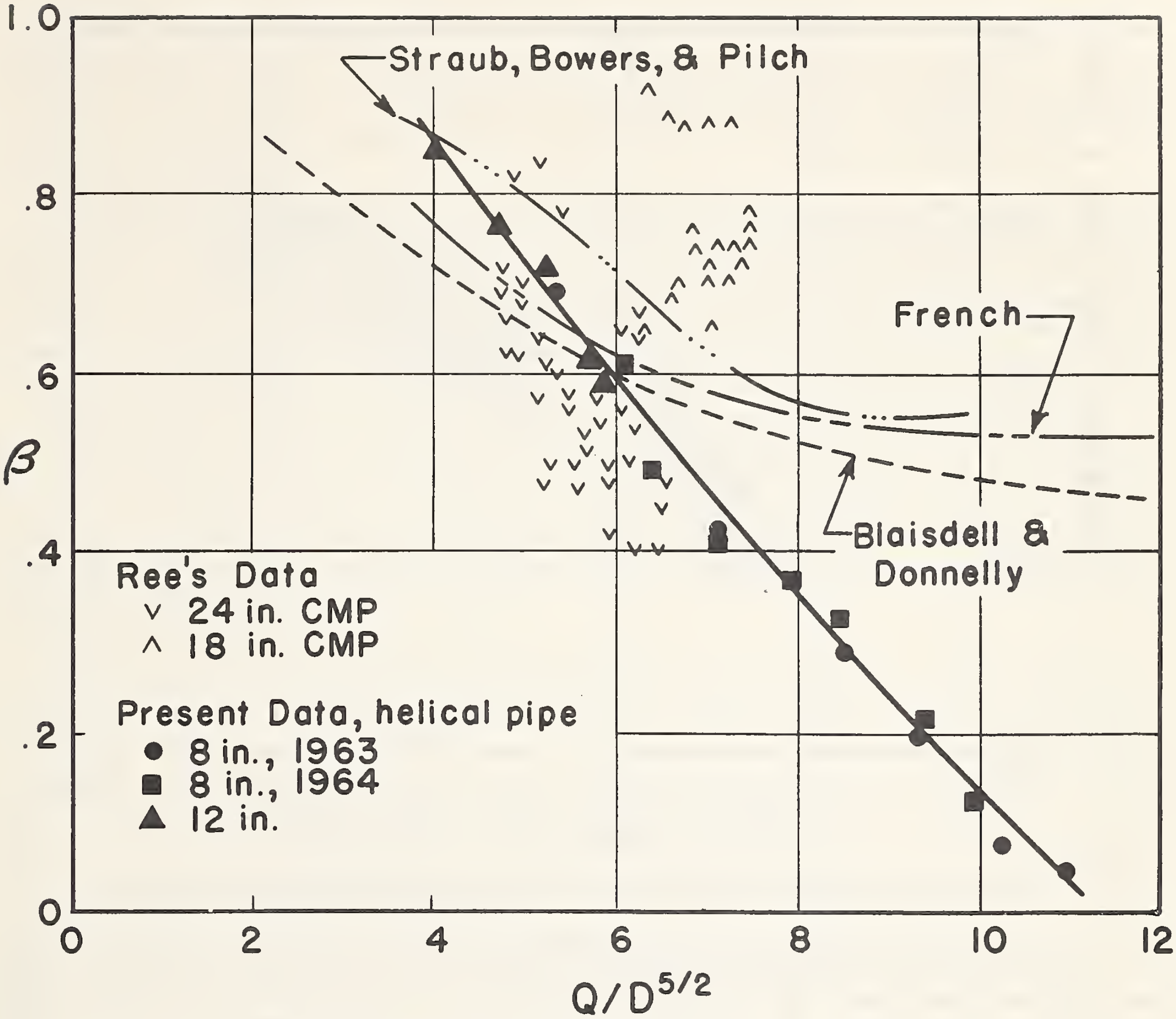

Figure 5.--Position of hydraulic grade line at the pipe outlet.

DISCUSSION

The Darcy-Weisbach f-values from the Stillwater tests of the 12-inch-diameter pipe range from 0.038 to 0.046. This compares with an f-value range of 0.038 to 0.044 reported in the Handbook (1) and an f-value of 0.040 reported by Chamberlain (4). The f-values from the Stillwater tests for the 8-inch pipe range from 0.035 to 0.043. This compares with an f-value range of 0.033 to 0.038 reported in the Handbook of Drainage and Construction Products (1).

A comparison of the friction factor values computed from the Stillwater tests shows the larger pipe to have the larger friction factor. This agrees with the findings reported in the Handbook of Drainage and Construction Products (1). The Handbook f-value ranges plotted on figure 6 show an increase with pipe diameter. Since all sizes of spiral corrugated metal pipe are made from the same width of sheet, the helix angle ϕ increases with pipe diameter to a limit of 90 degrees. At 90 degrees, spiral corrugated pipe becomes ordinary corrugated pipe. For the small pipe sizes the flow follows the spiral corrugation giving a larger flow area than would be

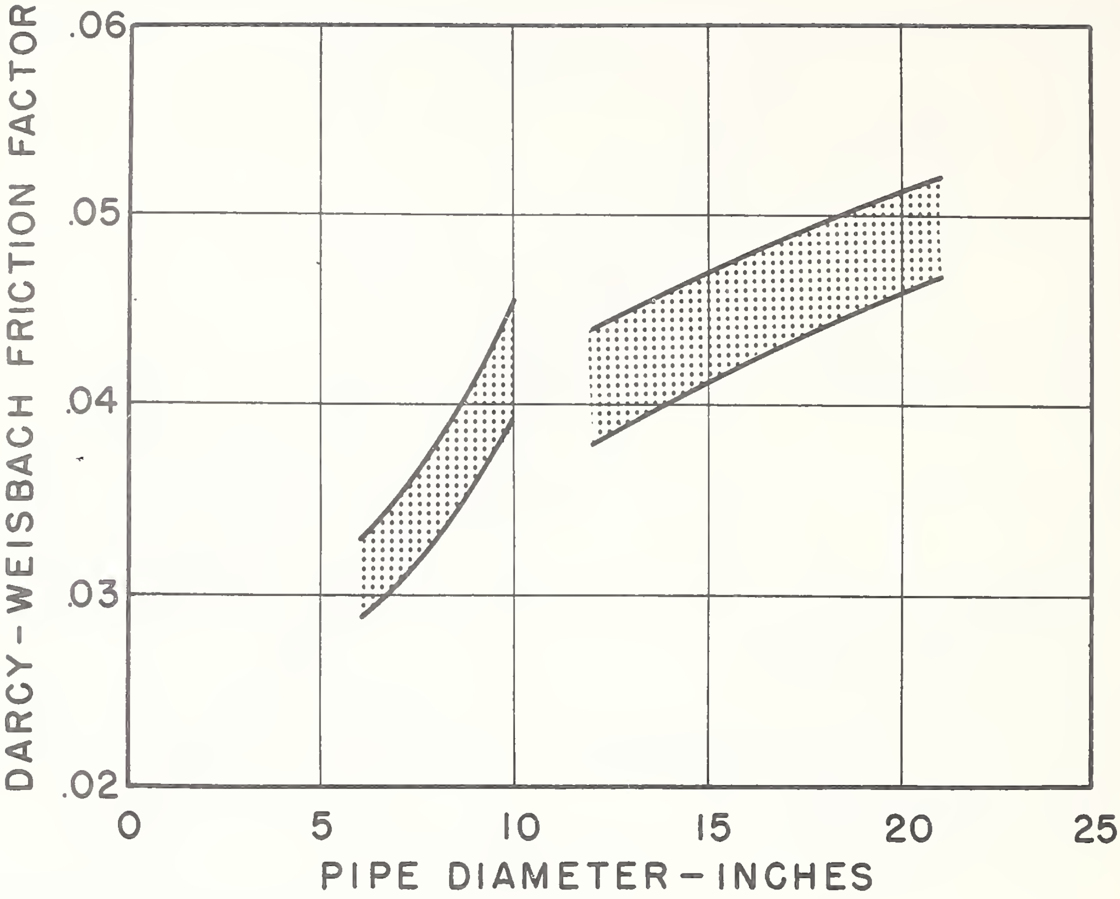

Figure 6.--Darcy-Weisbach friction factor value ranges for helical corrugated metal pipe (1).

the case if the flow had to go over the corrugations. The losses due to the longer spiral flow path are probably less than those caused by the flow over the transverse corrugations. As the pipe size increases, more flow probably occurs across the corrugations, with corresponding increases in friction losses. For large spiral pipes the friction losses should approach those for ordinary corrugated pipes having the same size corrugation.

The plottings on figure 6 represent two different corrugation forms; hence the discontinuity in the curve. For 10-inch and smaller spiral pipe the corrugations have a pitch of 1.5 inches and a depth of 0.25 inch while for 12-inch and larger spiral pipe these values are 2.0 inches and 0.50 inch, respectively. The depths are nominal values and do not agree exactly with those measured for the test pipe.

Friction factor values calculated from periphery pressure differ slightly from those calculated from the center pressure measurements. The differences range from about 1 percent at the higher Reynolds numbers to about 10 percent at the lower Reynolds numbers. To determine if these differences were due to a lack of precision in the experimental method, a study was made to determine the possible error (percent) in each of the terms in equation 2. The sum of these errors, is the possible error (percent) in the determination of the friction factor.

Equation 2 shows that the friction factor is a function of S, the slope of the hydraulic grade line; D, the pipe diameter; and h_v, the velocity head of flow in the pipe. The nominal diameter was employed in the calculations for both sets of measurements. Therefore, this term is without error for the purposes of this comparison. Similarly, the velocity head term is without error since the same discharge and same diameter were used for both calculations. Therefore, the only error entering into the determination of the difference between the two friction factor values is the error in the slope of the hydraulic grade line.

The slope of the hydraulic grade line for both the center pressure and periphery pressure measurements was calculated by the method of least squares. The linear regression technique used to calculate the slope values, permitted an analysis to determine the deviation of the slope. The ratio of the deviation of the slope to the calculated slope was calculated for each f value and this was assumed to represent the possible error in the slope term and, therefore, the possible error in the f value.

The error in the f values varied from about 2 percent to about 9 percent for Reynolds numbers less than 1.3 X 10^5. Although error values at the low Reynolds numbers were somewhat erratic, there was a general trend for the error to decrease as the Reynolds number increased. For Reynolds numbers greater than 1.3 X 10^5, the error in the f value was fairly constant in a range from about 1/2 percent to about 2-1/2 percent. The errors for the f values calculated from the center pressure measurements were slightly less than the errors for the f values calculated from the periphery pressure measurements.

This analysis showed that the error in the f values could range from about 1 percent to about 9 percent because of the lack of precision in measuring the slope of the hydraulic grade line. This possible error is about the same as the difference between the two f values calculated from the two different points of pressure measurement. The differences in the two sets of f values are probably not real, but are due to the lack of precision in obtaining the pressure measurement at the two locations in the pipe. Thus, for determining the hydraulic grade line, peripheral measurements are as satisfactory as the center measurements and are easier and more economical to obtain.

The two series of tests run on the 8-inch helical pipe, conducted about 1 year apart, permitted a comparison to determine if, and how much, the surface roughness changed in the 1-year period. The f-values are compared in figure 7 for the free outfall tests only, since for these tests the exit conditions were identical. It is seen that a small, but real, difference does exist between the f-values computed for the two test series. This difference is about 2-1/2 percent with the higher friction loss associated with the older condition. Apparently slight oxidation of the pipe surface caused the roughness increase. The pipe had not been used for the time intervening between the two test series except for a few short demonstration runs, so the roughness changes cannot be attributed to the effect of flows.

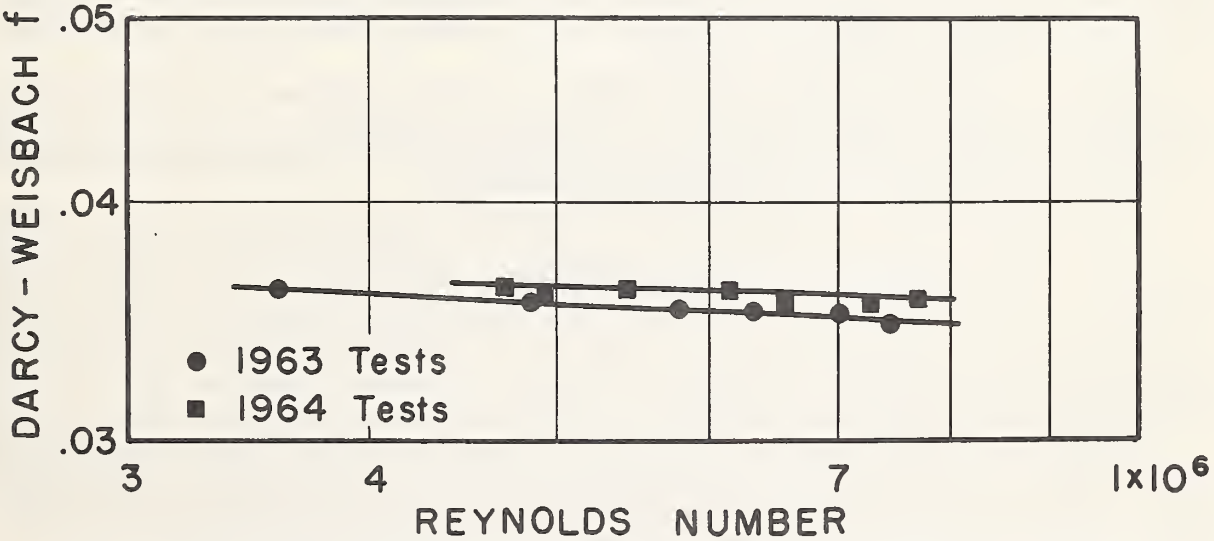

Figure 7.--Comparison of the 1963 and 1964 friction factor values for 8-inch helical corrugated metal pipe with free outfall condition.

The outlet restriction on the 8-inch-diameter pipe for the 1964 tests was changed from that used in the earlier tests to determine if the kind of restriction affected the resultant friction values. That it might have some effect seemed probable from the results of the 12-inch pipe tests, which showed higher f-values for the restricted flows than for the free outfall flows (see fig. 4a). Reasoning that the restriction caused the increase by inhibiting spiral motion of the outflow, a new kind of restriction (shown on figure 2 as "alternate outlet for 8-inch-diameter pipe") not having this characteristic was used on the 8-inch pipe for the first series of tests. The result showed no difference between friction factor values for the restricted or free outfall flows (see fig. 4b). This seemed to vindicate the judgment that the form of the restriction did affect the friction loss. However, it seemed unwise to conclude this by comparing the results on the 12-inch pipe with the results on the 8-inch pipe. Therefore, it was decided to retest the 8-inch pipe with the same type of outlet used on the 12-inch pipe--the simple vertical, sliding gate at the end of the pipe.

The friction factors obtained with the two different restrictions cannot be compared directly since a change in the pipe had occurred. Thus, it is necessary to compare the f-values for the restricted condition with those for free outfall separately for each test series. The f-values for the 8-inch pipe tests portrayed on figures 4b and 4c show that the use of a more restrictive outlet did result in slightly higher f-values for flows at the higher Reynolds numbers. However, this increase is so small that for design purposes for conditions of like velocity it can be assumed that no difference exists.

The average Manning n-value for the 12-inch-diameter pipe found at Stillwater, 0.015, compares well with the average n-value of 0.016 computed from the Handbook f-values (1). For the 8-inch-diameter pipe the average Stillwater n-value is identical to the average Handbook n-value, 0.013.

The Manning n-values presented in tables 1, 2, and 3 for the 12-inch and 8-inch pipes were computed using equation 4, which is based on full pipe flow in this present analysis. Therefore, these Manning n-values are applicable only for full pipe flow and should not be used for flow analysis of pipes flowing part full.

The values of β given in tables 1, 2, and 3 were determined from a hydraulic grade line based on pressure measurements taken very near the center of the pipe cross section. Pressures were also measured at the periphery of the pipe. The center pressure determinations were used to calculate β values for comparison with β values obtained by other experimenters. The comparison is made on figure 5.

Values of β from this experiment show good agreement with other data in the range of lower flows, but they depart from the others in the range of the higher flows. The relationship of β to discharge for the helical pipe does not level off at the mid-range but instead it continues a well-defined trend of decreasing β with increasing discharge.

An observation that can be made in the comparison is that the physical nature of the pipe itself may change the flow pattern and might account for the β differences. The curves on figure 5 by Blaisdell and Donnelly (3) and French (5) were obtained from tests on smooth lucite pipes, whereas the curve of Straub, Bowers, and Pilch (7) was obtained from tests on 24-inch and 36-inch-diameter concrete pipes. The other data are for standard corrugated metal pipe[4] and helical corrugated metal pipe. Apparently the roughness or physical form of the conduit as well as the discharge determines the value of β.

β values for the 8-inch-diameter helical pipe computed using the periphery pressures are shown in figure 8 with the β values computed using the center pressures. Both sets of β values show a downward trend with increasing discharge. The values obtained from the peripheral pressure measurements are higher than those based on the center measurements.

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[4] Ree, W. O. Hydraulics tests on a pipe outlet spillway. Unpublished material.

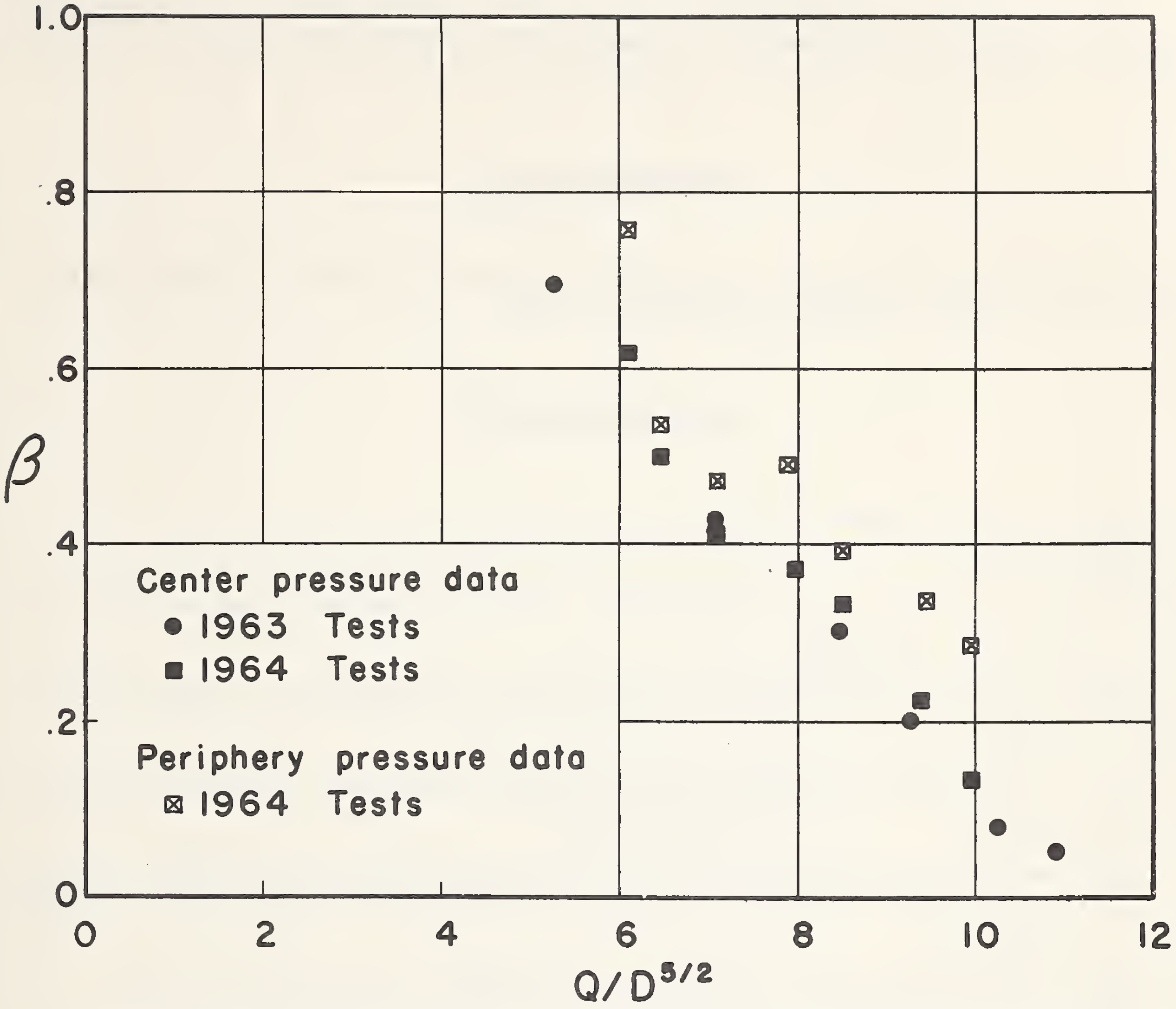

Figure 8.--Position of hydraulic grade line at the outlet of the 8-inch helical corrugated metal pipe with free outfall condition.

CONCLUSIONS

The results of the present study showed that the Darcy-Weisbach friction factor f for the 12-inch helical pipe had no systematic variation with the Reynolds number in the flow range studied, but the friction factor f-values for the 8-inch helical pipe decreased slightly with increasing Reynolds number. For the 12-inch pipe the f-value varies from about 0.040 to about 0.042 while for the 8-inch pipe the f-value decreases from about 0.042 at a Reynolds number of 5.0×10^4 to about 0.034 at a Reynolds number of 7.0×10^5.

The Manning n-values were nearly constant for each pipe. Therefore, the average n-values of 0.015 and 0.013, respectively, for the 12-inch and 8-inch-diameter helical pipes should be adequate for most design purposes. These values apply for full pipe flow only and should not be used for pipes flowing part full.

The position of the hydraulic grade line at the exit of a conduit flowing full and discharging freely is dependent on the discharge, and possibly on the physical form of the pipe. More information is needed to accurately determine its position for different forms of pipes.

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LITERATURE CITED

(1) Armco Drainage and Metal Products, Inc.
1958. Handbook of drainage and construction products, inc. 529 pp., illus., Chicago
(2) Blaisdell, F. W.
1960. Hood inlet for closed conduit spillways. Amer. Soc. Civ. Engin., Hydraul. Div. Jour., 86 (HY 5): 7-31, illus.
(3) Blaisdell, F. W., and Donnelly, C. A.
1958. Hydraulics of closed conduit spillways, part X, the hood inlet. St. Anthony Falls Hydraul. Lab., Tech. Paper 20, Ser. B, 41 pp., illus.
(4) Chamberlain, A. R.
1960. A discussion of friction factors in corrugated metal pipe. Amer. Soc. Civ. Engin., Hydraul. Div. Jour., 86 (HY 3): 67-72, illus.
(5) French, John L.
1956. Second progress report on hydraulics of culverts; pressure and resistance characteristics of a model pipe culvert, Natl. Bur. Standards, U.S. Dept. Com., Processed, 35 pp., illus.
(6) Rouse, Hunter, ed.
1950. Engineering Hydraulics. 1039 pp., illus. John Wiley and Sons, Inc., New York
(7) Straub, L. G., Bowers, C. E., and Pilch, M.
1958. Resistance to flow in two types of concrete pipes, St. Anthony Falls Hydraul. Lab., Tech. Paper 22, Ser. B, 148 pp., illus.